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Co-Orbiting Scroll Design and Operational Characteristics

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ABSTRACT

The drive systems of all scroll compressors in commercial production are of a conventional fixed-orbiting arrangement. This is mechanically simple and well within current mechanical design practice. For radially compliant designs it also has the disadvantage of generating large variations in inertial flank forces in variable speed or large capacity applications.

Another radially compliant drive arrangement is the co-orbiting drive, in which both scrolls are allowed to orbit with independent radial constraints. The wrap contact force becomes a virtual function of gas forces only and may be controlled within narrow limits over a wide range of compressor speed or size.

INTRODUCTION

The fixed-orbiting scroll drive arrangement in which one scroll is held stationary in the radial direction and the other is driven in an orbital path around its center is very common in practice today. It has the advantage of being mechanically simple, generally well understood, and falls well within the parameters of current mechanical design practice. For radially compliant designs whose flank loads are subject to the inertial force of the orbiting scroll, it also has the disadvantage of generating large variations in inertial forces in variable speed or large capacity applications.

Beyond a certain speed range, either high or low, special measures must be taken to assure good sealing at low speed where inertial forces are low and manageable flank loads at high speed where the forces are high. Above a given capacity range, where the orbiting scroll may become large and massive, similar measures must be taken to deal with very high inertial loads carried by the flanks. Effective radial compliance becomes nearly impossible and many have used fixed a crank radius or limited compliance for these extreme applications.

Throughout the history of scroll compressor development, other drive arrangements which provide the relative orbital motion required by the scroll concept have been known but less well developed. For various reasons, activity in recent years has addressed a co-rotating drive in which both scrolls rotate about offset axes, eliminating the high radial inertial forces associated with orbital motion. Leaving behind the special set of bearing and sealing problems this approach entails, radial compliance is possible here but has a large added degree of design complexity and is often omitted.

Another radially compliant drive arrangement available to designers is that of the co-orbiting drive, in which both scrolls are allowed to orbit with independent radial constraints. One scroll is driven by the drive train while the motion of the other is determined by geometry and gas forces. Radial inertial forces are reacted through the drive shaft and compressor frame instead of through the scroll wraps. The wrap contact force becomes a function of gas forces only and may be controlled within narrow limits over a wide range of compressor speed or size.

BACKGROUND

The earliest known example of a co-orbiting scroll drive arrangement goes back to the U.S. Patent issued to Léon Creux in 1905 [1]. In his reversible steam expander, he would clutch the mechanism between co-rotating and co-orbiting modes to achieve the reversing feature. Interestingly, this earliest reference does not make use of a conventional fixed-orbiting arrangement at all! The two drive modes shown instead, co-orbiting and co-rotating, are the two which have come under more recent development. However, Creux's co-orbiting drive arrangement was constructed around two fixed radius cranks as part of a single solid shaft and no radial compliance was envisioned for the device.

The co-orbiting drive in its current form was first disclosed in 1975 by Niels O. Young [2]. He visualized an arrangement where the driven scroll was connected to a shaft with a fixed crank throw, much like a non-radially compliant fixed-orbiting drive arrangement except that now the fixed radius was larger than the built-in orbit radius of the scroll set. The free (formerly fixed) scroll was given an independent radial restraint which allowed it to go through its own smaller, so-called minor orbit in response to gas pressures and mechanical flank loading. Young also pointed out the free scroll would now need a second independent anti-rotation device in the same manner as the driven scroll. The general object of this design was to control flank loading. That is the object of this study as well and we will look at the kinematics of this arrangement with an eye to the design of a practical, working compressor.

DRIVE KINEMATICS

We begin by looking at the arrangement of the scroll components as implied by the description of Young's co-orbiting design. We will proceed to analyze the forces acting on this arrangement and determine the operating characteristics from this analysis.

Component Orientation

Unlike conventional radially compliant designs which use a radially compliant driven scroll, the co-orbiting design has a fixed crank for the driven scroll that maintains a constant radius orbit path, much as in a noncompliant design. An antirotation coupling is used, as in conventional designs.

The free scroll, which in conventional designs remains fixed radially, is allowed to orbit. It's orbit path is limited to an arbitrary fixed radius. Where Young anticipated a radius larger than the orbit radius, we will begin by merely making it arbitrary. The free scroll is also constrained from rotating by the use of a second antirotation coupling.

With the appropriate selection of driven scroll crank radius R_c and free scroll minor orbit radius R_p along with the scroll profile's orbit radius R_{or}, the co-orbiting mechanism will become a radially compliant assembly that allows flank contact between scroll wraps. Any variation in orbit radius, whether due to part tolerance, liquid, or debris, will result in a variation of the phase angle between the driven and free scrolls. The flank contact force is dependant on gas loading within the scrolls and is virtually independent of the inertia effects of the orbiting components.

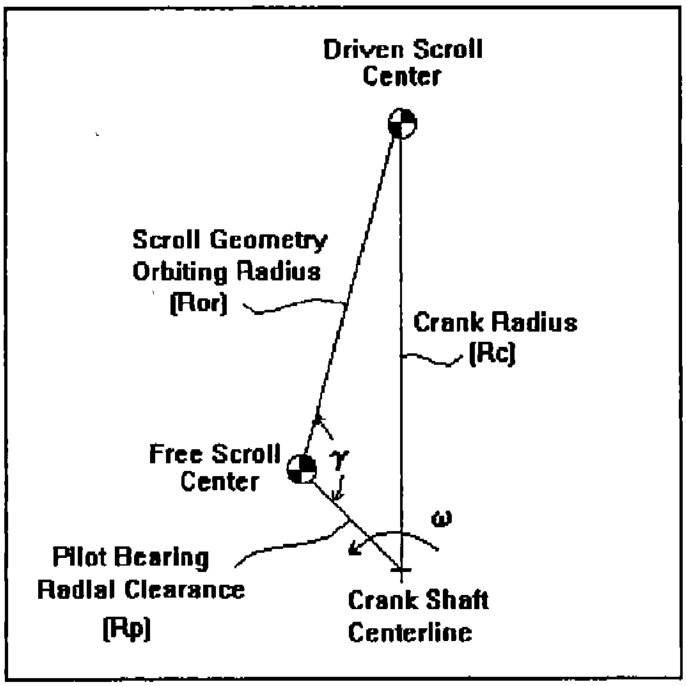

FIGURE 1
Drive Triangle

Drive Triangle

Figure 1 illustrates the orientation of this mechanism and is referred to as the drive triangle. The angle γ is referred to as the compliance angle.

FORCE VECTOR RELATIONSHIPS

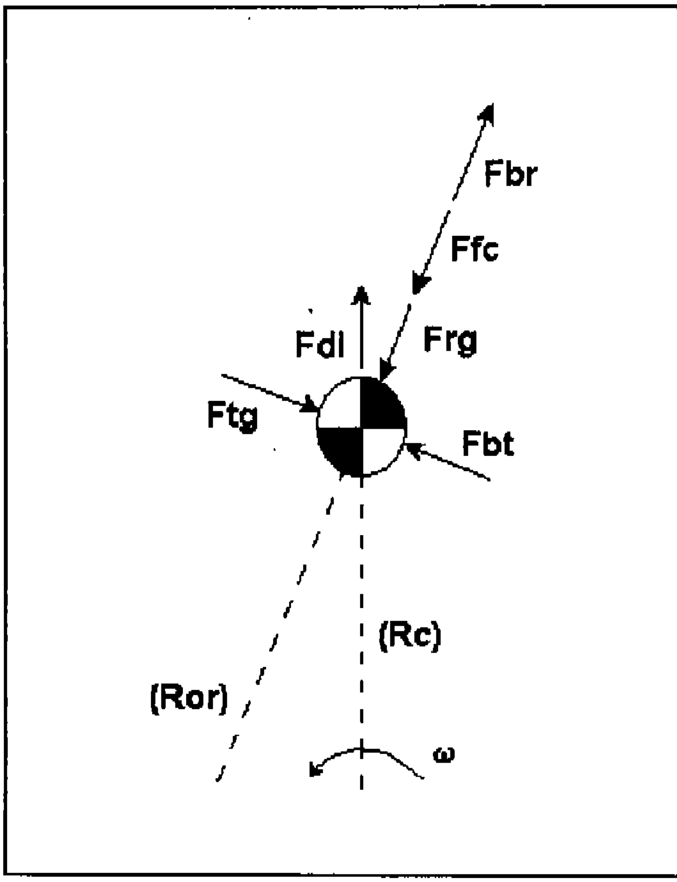

FIGURE 2
Driven Scroll
Free Body Diagram

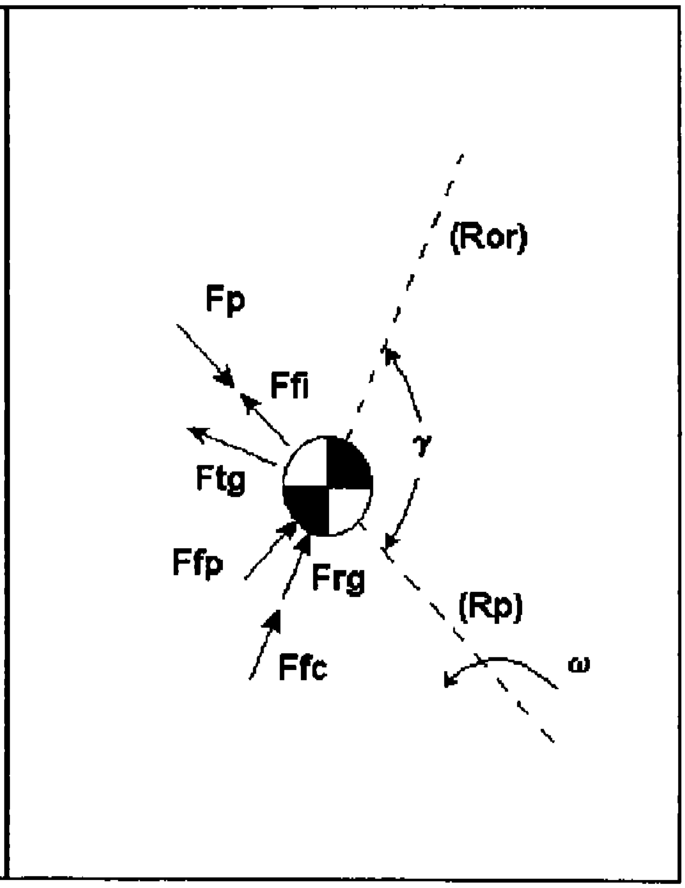

FIGURE 3
Free Scroll
Free Body Diagram

Figure 2 shows a free body diagram of the driven scroll. The gas force in the orbital plane is separated into tangential (F_{tg}) and radial (F_{rg}) components. The tangential component acts normal to the orbit radius vector of the scrolls R_{or} and the radial component acts parallel to R_{or} as shown. The flank contact force F_{fc} acts collinear to the radial gas force (which tries to separate the scrolls radially) and in the same direction. The driven scroll inertia force F_{di} acts in line with the crank radius. F_{br} and F_{bt} represent the resultant loads on the driven scroll bearing as radial and tangential components.

Figure 3 shows a free body diagram of the free scroll. The gas forces are again shown as F_{tg} and F_{rg}. The free scroll inertia force F_{fi} and the radial free scroll bearing reaction force F_p act in line and through the centers of the crank shaft and free scroll. The radial free scroll bearing frictional force F_{fp} acts normal to F_p in a direction to oppose free scroll motion and, assuming coulombic friction, is equal to F_p times the pilot bearing frictional coefficient, μ.

The free scroll bearing frictional force F_{fp} is included here because it is the only force acting in its particular direction and, as we will see, influences the flank loading in the co-orbiting drive. Other frictional forces are neglected since as a practical matter they may be rolled into or included with other forces acting in the same direction, such as F_{tg} for flank friction, in a more rigorous analysis than we are able to present here.

At a given operating condition and speed, all the forces acting on the free scroll are known except for F_p and F_{fc}.

The resultant of the forces acting on the free scroll try to drive the scroll outward from the co-orbiting drive triangle. The coaction between the free scroll radial bearing and the free scroll partially counter this force. In a not quite normal direction, the flank contact with the driven scroll serves to complete the mechanical restraint of the free scroll. The following force equations are used to determine the magnitude of free scroll bearing and flank contact forces.

Summing the forces in the direction of the free scroll radial bearing reaction and setting them to zero results in

$$\Sigma\ F = 0 = F_{FI} - F_{P} - F_{RG}COS\gamma - F_{FC}COS\gamma + F_{TG}SIN\gamma \tag{1}$$

Solving equation (1) for the pilot bearing force gives us

$$F_{P} = F_{FI} - F_{RG}COS\gamma - F_{FC}COS\gamma + F_{TG}SIN\gamma \tag{2}$$

Summing the forces in the normal direction results in

$$\Sigma\ F = 0 = F_{FP} + F_{RG}SIN\gamma + F_{FC}SIN\gamma + F_{TG}COS\gamma \tag{3}$$

Combining equations (2) and (3) and solving for the flank contact force gives us

$$F_{FC} = \frac{F_{FI}\mu + F_{TG}(\mu SIN\gamma + COS\gamma)}{\mu COS\gamma - SIN\gamma} - F_{RG} \tag{4}$$

or, neglecting friction,

$$F_{FC} = -\frac{F_{TG}COS\gamma}{SIN\gamma} - F_{RG} \tag{5}$$

Equation (4) shows the only effect that inertial forces have on the flank contact force is dependant on the friction of the free scroll radial restraint bearing. Equation (5) states that if this frictional factor can be made characteristically low, as in a hydrodynamic bearing for instance, the flank contact force becomes virtually independent of compressor operating speed as a practical matter. The terms containing the tangential and radial gas forces will predominate and the flank load will vary with operating condition only.

The negative sign in the main term of Equation (5) also shows that the value of γ must lie in the second or fourth quadrant for the ratio of cosine to sine to be negative and yield a positive net flank load. Additionally, γ must be somewhat above the nominal minimum values to overcome the effect of F_{rg} in the second term. When we consider friction as in Equation 4 the practical minimum value of gamma will be somewhat larger than even this. On the other hand, selection of gamma near the upper end of the respective ranges will result in very high flank forces. Selection of an appropriate compliance angle must be done with care.

The case illustrated in Figure 1, where γ lies in the second quadrant, is called the "leading" configuration since the free scroll leads the driven scroll in its orbit. The free scroll bearing reaction acts inward as shown in Figure 3 and all the free scroll requires is a radial restraint, such as a pilot bearing, as illustrated in Young's patent, to prevent it from moving outward from its minor orbit path.

The drive triangle for the case where γ is in the fourth quadrant is shown in Figure 4. This is the "lagging" case, since the free scroll follows the driven scroll in its orbit. The bearing reaction force on the free scroll now acts outward since the free scroll now has an inclination to move inward. A pilot bearing is now insufficient and some other bearing, such as a rotating link, must be used to restrain the free scroll to a circular orbit.

There may be individual design factors for selecting either a leading or lagging compliance triangle as they are integrated into the overall compressor layout. Either way, both types of co-orbiting drives can result in a suitable radial compliance design with similar flank loading characteristics.

Careful selection of crank and minor orbit radii will both limit the free scroll inertial forces, which influence flank load through friction effects, and set the flank contact forces within an easily manageable range. It is very interesting to note that equations (4) and (5) predict this radial compliance concept will provide positive and controlled flank sealing at virtually any operating speed, from the highest practical speed as limited by flow restriction or counterweight forces and down to speeds as low as theoretically zero velocity.

Having solved for the flank contact force, the force summation can also be performed for the driven scroll free body diagram to determine the net drive bearing load for design purposes.

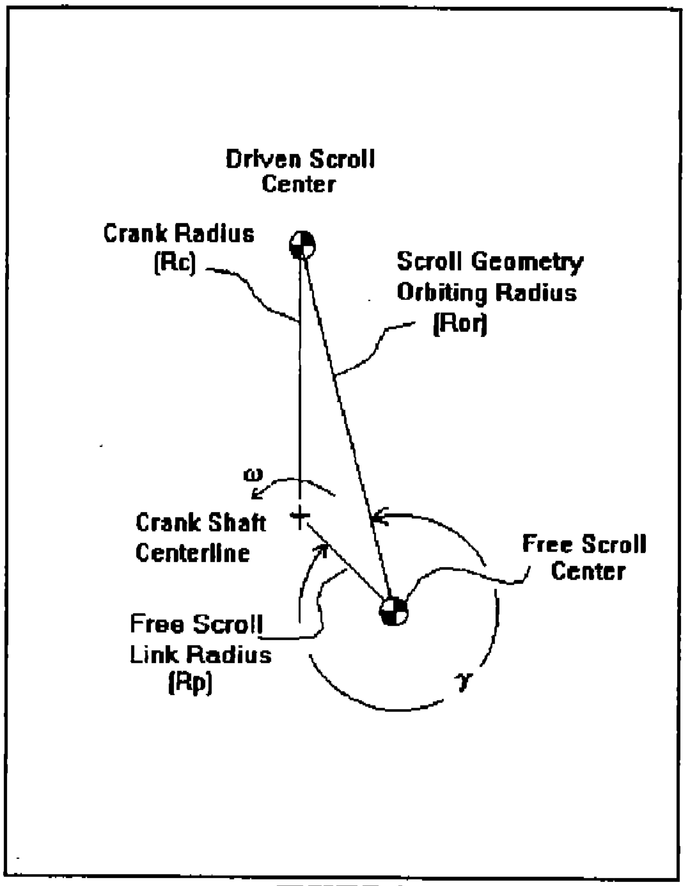

FIGURE 4
Drive Triangle
Lagging Case

DESIGN EXAMPLE

One application of this design might be in large displacement compressors with fairly massive orbiting components which otherwise would require either counterbalanced radial compliance drive linkages or a noncompliant design. In this example, we look at a large scroll designed to deliver 15 tons cooling capacity at 50 Hz operation (2900 rpm) with R134a, but driven at 60 Hz. The orbiting scroll is iron and weighs around 12.5 pounds.

	Flank Loading, lbf	
Condition	Conventional Compliance	Co-Orbiting Compliance
Refrigeration	1,249	105
Air Conditioning	1,202	267
Maximum Load	1,158	356

If we now drive this same compressor at variable speed at, say, an air conditioning load, we see loads

	Flank Loading, lbf	
Operating Speed	Conventional Compliance	Co-Orbiting Compliance
30 Hz	223	267
60 Hz	1,202	"
90 Hz	2,833	"
120 Hz	5,117	"

CONCLUSION

Although the co-orbiting drive concept has been around for some time, very little attention has been paid to it in light of the successes of compliant and noncompliant fixed/orbiting designs. However, co-orbiting scroll drives do have some unique operating characteristics which make them worthy of consideration for variable speed applications as well as large displacement machines where scroll mass begins to become a limiting factor. The kinematics of operation are fairly straightforward and stable. Other portions of the design which may seem as unconventional to the scroll compressor engineer as the drive concept itself are the free scroll radial bearing provision and the need for dual anti-rotation devices. The potential for providing controlled radial compliance under extreme conditions promise to extend the application of scrolls beyond the limits of practice today.

LIST OF SYMBOLS

γ	Compliance angle
μ	Coefficient of sliding friction
ω	Shaft angular velocity
Fbr	Drive bearing radial reaction force
Fbt	Drive bearing tangential reaction force
Fdi	Driven scroll inertia force
Ffc	Flank contact force
Ffi	Free scroll inertia force
Ffp	Free scroll radial bearing friction force
Fp	Free scroll radial bearing force
Frg	Radial gas force
Ftg	Tangential gas force
Rc	Crank Radius
Ror	Scroll orbiting Radius
Rp	Minor orbit radius (free scroll)

ACKNOWLEDGEMENT

We would like to recognize the contribution of David K. Haller to much of the analysis and development of the design techniques and relations presented here in part. Though we probably don't need to, we would admonish him to "always watch the numbers."

REFERENCES

[1] Creux, Léon, U.S. Patent 801,182, "ROTARY ENGINE," October 3, 1905.

[2] Young, Niels O., U.S. Patent 3,874,827, "POSITIVE DISPLACEMENT SCROLL APPARATUS WITH AXIALLY RADIALLY COMPLIANT SCROLL MEMBER," April 1, 1975.